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# Full-Scale Transmission Testing to Evaluate Advanced Lubricants

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The goal of the optimum oil phase is to develop a lubricant specially formulated for gearboxes. The lubricant is to have greater load-carrying capacity (about twice that of MIL-L-23699) and improved corrosion inhibiting properties. Candidates for the optimum oil are being developed and evaluated in laboratory and bench testing by oil companies and NAPC. The advanced oil phase (at least a decade or two away) is intended for future aircraft in which the transmissions would operate at high temperature and high loading.

NASA Lewis Research Center and the U.S. Army Propulsion Directorate, with their unique gear and transmission facilities, are supporting the Advanced Lubricants Program. The effect of the lubricant on the gear pitting fatigue life is being studied using the gear fatigue test apparatus at NASA Lewis. It was found that higher viscosity lubricants (compared to MIL-L-23699) with additive packages increased gear life (Townsend and Shimski, 1991).

In addition to the gear tests, full-scale transmission tests were performed in the NASA Lewis 500-hp helicopter transmission test stand. The Army OH-58A helicopter main-rotor transmission was used for the tests. The objective of the research reported herein was

(1) To develop an experimental testing procedure to simulate transmission failures experienced by the Navy in the field. These failures included planet bearing and sun gear fatigue, mast shaft ball-bearing micropitting, and spiral bevel gear scoring, all of which are related to the lubricant.

(2) To test to failure the above-mentioned components in the OH-58A transmission using a MIL-L-23699 based reference oil and then using the candidate, improved lubricants to demonstrate improved performance.

## APPARATUS AND TEST PROCEDURE

### OH-58A Main-Rotor Transmission

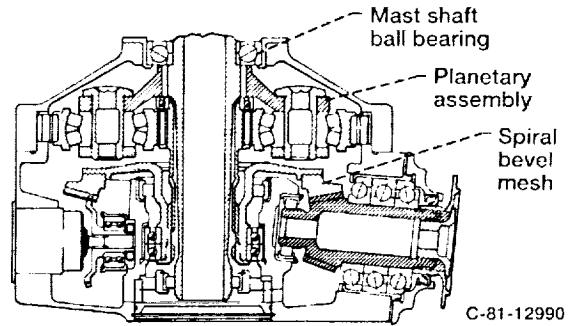
The OH-58A is a single-engine, land-based, light, observation helicopter. The helicopter serves both military (OH-58 Kiowa) and commercial (Bell Model 206 Jet Ranger) needs. The design maximum torque and speed for the OH-58A main-rotor transmission (Fig. 1) is 350 N-m (3100 in.-lb) input torque and 6060 rpm input speed (Warren and Young, 1969). This corresponds to 222 kW (298 hp). The transmission is a two-stage reduction gearbox with an overall reduction ratio of 17.44:1. The first stage is a spiral bevel gear set with a 19-tooth pinion that meshes with a 71-tooth gear. Triplex ball bearings and one roller bearing support the bevel-pinion shaft. Duplex ball bearings and one roller bearing support the bevel-gear shaft in an overhung configuration.

A planetary mesh provides the second reduction stage. The bevel-gear shaft is splined to a sun gear shaft. Both a three-planet (OH-58A) and four-planet (OH-58C) assembly were used for the tests. For the three-planet assembly, the 27-tooth sun gear drives three 35-tooth planet gears. The planet gears mesh with a 99-tooth fixed ring gear splined to the transmission housing. The planet gears are supported by double-row spherical roller bearings attached to the planet carrier. Power is taken out through the planet carrier splined to the output mast shaft. The output shaft is supported on top by a split-inner-race ball bearing and on bottom by a roller bearing. The four-planet assembly is different from the three-planet in that it has one more planet, the planet bearings are cylindrical rollers rather than spherical, and the planets are straddle mounted by the carrier rather than overhung. The four-planet assembly has significantly higher load-carrying capacity than the three planet.

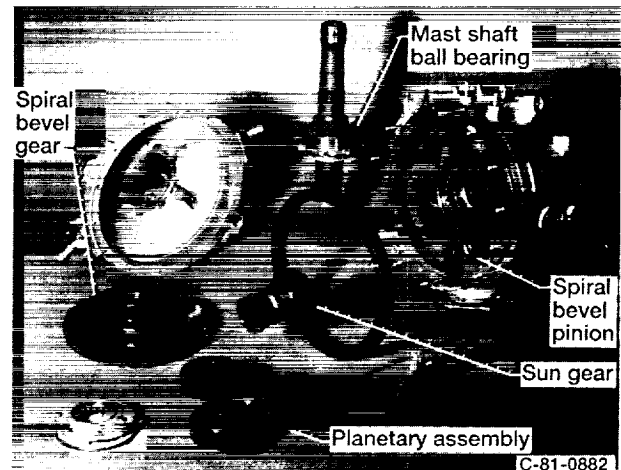
The 71-tooth bevel gear also drives a 27-tooth accessory gear. The accessory gear runs an oil pump, which supplies lubrication through jets and passageways located in the transmission housing.

### NASA 500-HP Helicopter Transmission Test Stand

The OH-58A transmission was tested in the NASA Lewis 500-hp helicopter transmission test stand (Fig. 2). The test stand operates on the closed-loop or torque-regenerative principle.



(a) Cross-sectional schematic.



(b) Disassembled view.

FIG. 1.—OH-58A helicopter main rotor transmission.

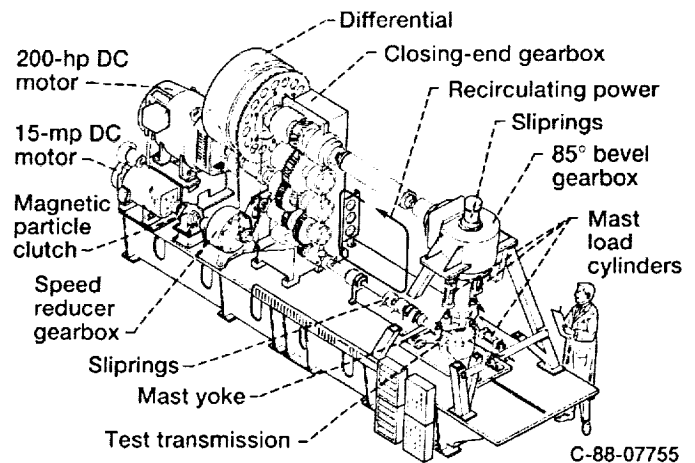


FIG. 2.—NASA Lewis 500-hp helicopter transmission test stand.

Mechanical power recirculates through a closed loop of gears and shafting, one of which is the test transmission. The output of the test transmission attaches to the bevel gearbox. The output shaft of the bevel gearbox passes through a hollow shaft in the closing-end gearbox and connects to the differential gearbox. The output of the differential attaches to the hollow shaft in the closing-end gearbox. The output of the closing-end gearbox connects to the input of the test transmission, thereby closing the loop.

A 149-kW (200-hp) variable-speed direct-current (d.c.) motor powers the test stand and controls the speed. The motor output attaches to the closing-end gearbox. The motor replenishes only losses due to friction since power recirculates around the loop.

An 11-kW (15-hp) d.c. motor provides the torque in the closed loop. The motor drives a magnetic particle clutch. For the OH-58A application, the clutch output does not turn but exerts a torque. This torque transfers through a speed reducer gearbox and a chain drive to a large sprocket on the differential gearbox. The torque on the sprocket puts a torque in the closed loop by displacing the gear attached to the output shaft of the bevel gearbox with the gear connected to the input shaft of the closing-end gearbox. This is done within the differential gearbox through use of a compound planetary system where the planet carrier attaches to the sprocket housing. The magnitude of torque in the loop is adjusted by changing the electric field strength of the magnetic particle clutch.

A mast shaft loading system in the test stand simulates rotor loads imposed on the OH-58A transmission output mast shaft. The OH-58A transmission output mast shaft connects to a loading yoke. Two vertical load cylinders connected to the yoke produce lift loads. A single, horizontal load cylinder, connected to the yoke, produces shear loads. A 13 790-kPa (2000-psig) gas nitrogen system powers the cylinders. Pressure regulators connected to the nitrogen supply of each of the load cylinders adjust the magnitude of lift and shear forces.

The test transmission input and output shafts have speed sensors, torquemeters, and slings. All three load cylinders on the mast yoke are mounted to load cells. The test transmission internal oil pump supplies lubrication. An external oil-water heat exchanger cools the test transmission oil. The 149-kW (200-hp) motor has a speed sensor and a torquemeter. The magnetic particle clutch has

speed sensors on the input and output shafts and thermocouples. A facility oil-pumping and -cooling system lubricates the differential gearbox, the closing-end gearbox, and the bevel gearbox. The facility gearboxes have accelerometers, thermocouples, and chip detectors for health and condition monitoring.

#### Testing Procedure

The component failures of interest for these tests were sun gear and planet bearing fatigue, mast shaft ball-bearing micropitting, and spiral bevel gear scoring. Ten tests were performed (Table I), accumulating over 900 hr of total run time, including test time and warmup time.

The OH-58A main-rotor transmission used for the testing was from the Corpus Christi Army Depot (CCAD) where it had been overhauled after 1908 hr run time (load history unknown). Only the spiral-bevel-pinion triplex ball bearing and the spiral bevel gear duplex ball bearing were replaced during the overhaul. All other transmission components had 1908 hr on them. Before running at NASA Lewis, the transmission was disassembled, inspected, instrumented, and reassembled. Thermocouples were installed on most bearing races. Slip rings on the transmission input and output shafts extracted the thermocouple signals from the bevel-pinion triplex and roller bearing inner races as well as the planet bearing inner races. Accelerometers were installed at various locations on the transmission housing in a manner similar to that reported by Lewicki and Coy (1987). The intent of the instrumentation was to provide an indication of a component failure during the tests.

Table I. Test operating conditions.

[Tests 2 to 10 at 6060 rpm transmission input speed, 1% oil flow rate to mast ball bearing.]

Test	Test time, hr	Transmission input torque, % of design max	Mast radial load, % of design max	Transmission inlet temperature, °C (°F)	Oil type	Mast bearing ball surface roughness, $\mu\text{m AA}$ ( $\mu\text{in. AA}$ )	Other	Component failures
1	71	---	---	---	MIL-L-23699 Brand A	---	Parametric studies of effect of speed, load, temp, and oil pressure on performance	None
2	100	100	110	82 (180)	MIL-L-23699 Brand A	---		Spiral bevel pinion spall
3	50	117	110	121 (250)	MIL-L-23699 Brand A	---		Sun gear spall, spiral bevel heavy wear/scoring
4	50	100	110	121 (250)	MIL-L-23699 Brand A	---		None
5	75	117	110	121 (250)	MIL-L-23699 Brand A	0.53 (21)		Two planet bearing spalls
6	100	117	110	121 (250)	MIL-L-23699 Brand A	0.56 (22)	40% oil flow rate to spiral bevel mesh & 4-planet system used for last 25 hrs of test	Planet bearing spall, sun gear spall
7	100	117	132	121 (250)	MIL-L-23699 Brand A	0.81 (32)	21% oil flow rate to spiral bevel mesh; reduced oil level; 4-planet system used	None
8	100	117	110	121 (250)	MIL-L-23699 Brand B	0.81 (32)	21% oil flow rate to spiral bevel mesh; reduced oil level	Two sun gear spalls/scoring
9	91	117	110	121 (250)	MIL-L-23699 Brand B	---	21% oil flow rate to spiral bevel mesh; reduced oil level	Planet bearing cage failure
10	100	117	110	121 (250)	DOD-L-85734	0.61 (24)	21% oil flow rate to spiral bevel mesh; reduced oil level	None

Test 1 consisted of parametric studies of the effect of speed, load, temperature, and oil pressure on OH-58A transmission performance. Information from an analysis performed on the OH-58A transmission (Lewicki et al., 1991, Appendix A) and the results from the parametric studies (Lewicki et al., 1991, Appendix B) were used to set the parameters expected to produce failure. Tests 2 to 7 concentrated on developing a testing procedure to fail the desired components using the MIL-L-23699 lubricant. (Lubricant properties are given in Table II.) The tests were performed sequentially, with the results of each test used to set the conditions of the next. Tests 8 and 9 concentrated on repeating the results from tests 2 to 7 but with a second commercial brand of MIL-L-23699 lubricant.

Test 10 was a repeat of test 8 but using a DOD-L-85734 lubricant. The physical and chemical properties of the MIL-L-23699 and DOD-L-85734 lubricants (viscosity, flash and pour points, acidity, etc.) are similar. The DOD-L-85734 lubricant has significantly higher load-carrying capacity as indicated by the Ryder Gear ratings (and somewhat by the four ball wear tests).

Table II. Measured properties of MIL-L-23699 and DOD-L-85734 lubricants.

Parameter	MIL-L-23699 Brand A	MIL-L-23699 Brand B	DOD-L-85734
Kinematic viscosity, cS, at — 38°C (100°F) 99°C (210°F)	27.02 5.03	26.47 5.08	28.15 5.35
Flash point, °C (°F)	280 (500)	280 (500)	254 (490)
Pour point, °C (°F)	-59 (-75)	-57 (-70)	-62 (-80)
Total acid number, mg KOH/g	0.00	0.37	0.40
Rubber compatibility*, H - type, swell, % F - type, swell, % Silicon, swell, % Silicon, tensile loss, %	12 13 15 9	16.1 15.3 8.6 16.3	10.5 13.1 0 55.3
Ryder gear load capacity, % reference oil	122	122	164
Four ball wear test, wear scar diameter, mm 10 kg 40 kg	0.569 0.867	0.251 0.424	0.239 0.394

\*MIL-L-23699 run at 121°C; DOD-L-85734 run at 110°C.

## RESULTS AND DISCUSSION

### Analytical and Experimental Parametric Studies

A bending and contact stress analysis using American Gear Manufacturers Association (AGMA) standards was performed on gears in the OH-58A transmission main power train. Based on the AGMA stress analysis, it was concluded that 262 kW (351 hp) was the highest load to operate the OH-58A transmission to promote planetary pitting fatigue failures but not tooth bending fatigue failures. A scoring analysis was performed on the spiral bevel gear mesh using a method by Drago (1988). For conditions up to 275 kW (369 hp) and 121 °C (250 °F) oil inlet temperature, a low probability of scoring was predicted. The planet bearing, planet gear, and mast shaft ball bearing were analyzed using methods of Klecker et al. (1982), Wang and Cheng (1981a, 1981b), and Hadden et al. (1981). These components had extremely low predicted film thicknesses, implying a high probability of surface related problems

and that the influence of the lubricant could be significant. The results of the experimental parametric studies gave trends of the effect of speed, torque, and oil temperature on transmission performance. A detailed description of these results is given in Appendixes A and B of Lewicki et al. (1991).

### Planetary Results

Some success was achieved in developing a testing procedure to produce sun gear and planet bearing fatigue failures. A total of three planet bearing fatigue failures (along with one planet bearing cage failure) and four sun gear failures were produced using MIL-L-23699 lubricants (Fig. 3).

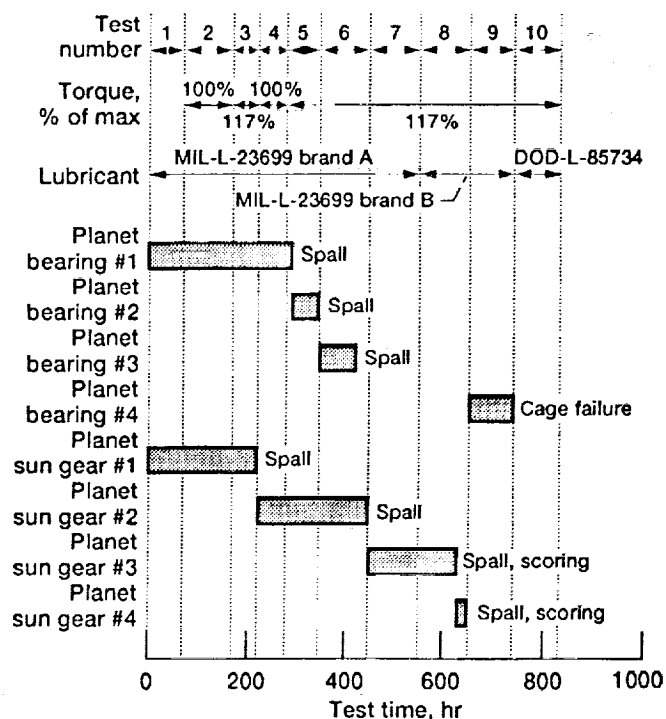


FIG. 3.—Sun gear and planet bearing failures.

**Planet bearing failures.** For MIL-L-23699 lubricant brand A, three planet bearing fatigue failures occurred. The first occurred on one of the original planet gears of the transmission during test 5 (Fig. 4). The failure was a fatigue spall on the inner race. This component acquired 66 hr of run time at operating conditions of

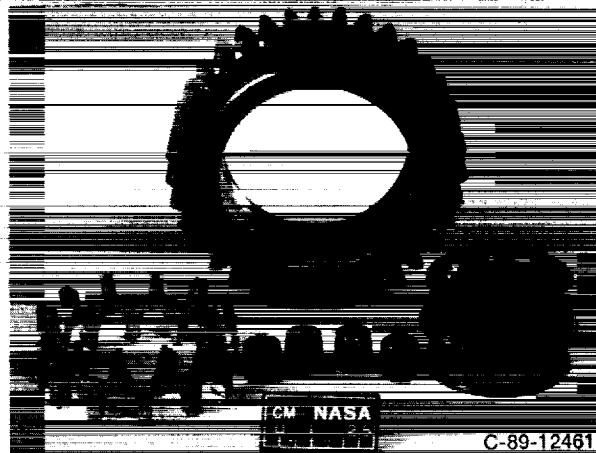


FIG. 4.—Planet bearing fatigue failure during test 5.

117-percent torque and 121 °C (250 °F) oil inlet temperature from tests 3 and 5. The second and third planet bearing fatigue failures occurred at these operating conditions. One occurred after 59 hr of test 5, and the other after 75 hr of test 6. The failures of all three bearings were similar. The spalls covered both rows of each bearing and about one-third of their circumferences. The large amount of spalling indicates some run time occurred after initial spalling. All failures were detected from the conventional chip detector on the OH-58A transmission.

For MIL-L-23699 lubricant brand B, one planet bearing cage failure was produced after 91 hr of test 9 (Fig. 5). This failure was catastrophic in that the cage for one of the planet bearings was completely destroyed and the rollers and raceways showed excessive scoring and distress. This failure was detected by a sudden increase in component temperatures and vibration.

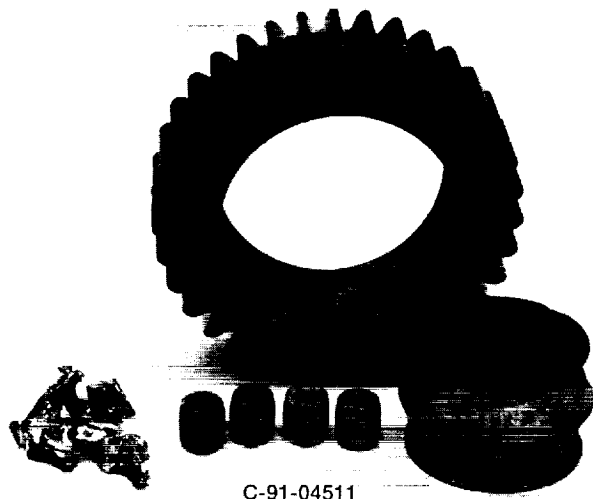


FIG. 5.—Planet bearing cage failure during test 9.

**Sun gear failures.** Two sun gear fatigue failures were produced using MIL-L-23699 lubricant brand A. The first occurred on the original sun gear of the transmission after test 3 (Fig. 6). A large fatigue spall occurred on one of its teeth covering nearly three-quarters of the face width and much of the tooth load zone. The second sun gear failure occurred after test 6. This failure was a small pit at the edge of one tooth.

Two sun gear fatigue failures occurred using the MIL-L-23699 lubricant brand B. The first was during test 8. This gear had spalls and some scoring on many of its teeth. The second sun gear failure occurred after running the last 20 hr of test 8. This gear again had spalls and some scoring on many of its teeth.

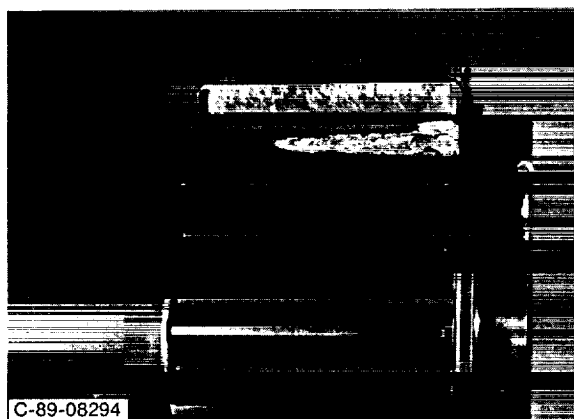


FIG. 6.—Sun gear fatigue failure after test 3.

**Additional comments.** The testing procedure using operating conditions of 117-percent torque and 121 °C oil inlet temperature seemed to produce planetary fatigue failures repeatably using the MIL-L-23699 lubricants. Traditionally, gear and bearing fatigue lives for components run under identical conditions occur with some scatter. Because of the relatively low number of failure data points (compared to conventional component fatigue testing), the confidence level of reproducing failures was not high. Note that the intent of the testing was not, however, to replace gear and bearing component fatigue testing. The intent was to run a transmission with its complex interactions between gears, bearings, and housing, and produce certain types of failures. Thus, from this viewpoint, success was achieved in developing a procedure to produce planetary fatigue failures.

The tests using the MIL-L-23699 lubricant brand A produced planet bearing inner race fatigue failures within about 75 hr when run at 117-percent torque and 121 °C oil inlet temperature. This procedure did not produce bearing fatigue failures when run on two successive 100-hr cases using the MIL-L-23699 lubricant brand B. It is not certain if the difference in results was due to oil differences. With regards to the sun gear, failures occurred at 117-percent torque and 121 °C oil inlet temperature. However, the results were ambiguous since components were run at different operating conditions. This was primarily done because the tests were performed sequentially, and the results from one test led to the conditions of the next.

As a final note, a single 100-hr run using a DOD-L-85734 lubricant (test 10) did not produce any component failures. This was encouraging in that no failures were produced using this advanced transmission lubricant. Because of the limited amount of data, however, these results by no means demonstrated that the fatigue life of the planetary system using a DOD-L-85734 lubricant was greater than that using a MIL-L-23699. Note that only one test using the DOD-L-85734 oil was performed (because of rig scheduling conflicts).

#### Mast Shaft Ball Bearing Results

Only slight success was achieved in developing a testing procedure to produce mast shaft ball-bearing micropitting failures. A total of six bearings were tested and only one resulted in micropitting. The first bearing was used in tests 1 to 4. After these tests, the bearing was in very good condition with no significant wear marks on the balls or raceways. Note that the bearing was run with a reduced oil flow rate of 1-percent of nominal, which had no effect on the results.

A second bearing was modified as follows. The balls of the bearing were roughened to increase their surface roughness and decrease the elastohydrodynamic film thickness to surface roughness ratio. The bearing was disassembled, and the balls were placed in a glass container, one at a time, with a mixture of 200 g of 20-grit silicon carbide and 6.35-mm (0.25-in.) steel balls. The container was tumbled at 300 rpm for 6 hr. This was repeated for each ball of the bearing. The surface roughness of each ball was measured using a Talysurf measuring system. Some scatter existed in the measured roughnesses between the various balls and also in the measurements for each ball. The average ball surface roughness was  $0.53 \pm 0.10 \mu\text{m AA}$  ( $21 \pm 4 \mu\text{in. AA}$ ), where  $0.10 \mu\text{m}$  ( $4 \mu\text{in.}$ ) was one standard deviation. The bearing was then assembled in the transmission and run 75 hr at 110-percent of design maximum radial load and at an oil inlet temperature of 121 °C (250 °F) (test 5). After running and upon inspection, the bearing exhibited micropitting on the inner race (Fig. 7). The bearing ball surface roughnesses were remeasured after running. The average ball surface roughness was reduced to  $0.09 \pm 0.02 \mu\text{m AA}$  ( $3.6 \pm 0.7 \mu\text{in. AA}$ ). Thus, the induced bearing roughness was reduced substantially after running the 75 hr.

This testing procedure was repeated with a third bearing. The average ball surface roughness after tumbling was  $0.56 \pm 0.13 \mu\text{m AA}$  ( $22 \pm 5 \mu\text{in. AA}$ ). The bearing was run a total of 100 hr (test 6) at the same operating conditions of the previous bearing that micropitted. After running and upon inspection, the bearing had a

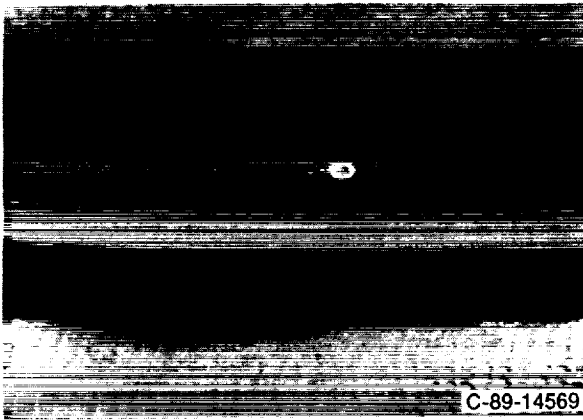


FIG. 7.—Mast shaft ball bearing inner-race micropitting after test 5.

noticeable wear mark along the load track of the inner race, but micropitting was not evident. The ball surface roughnesses were remeasured, and the average value was  $0.09 \pm 0.02 \mu\text{m AA}$  ( $3.7 \pm 0.6 \mu\text{in. AA}$ ). Thus, the average ball surface roughness results were nearly identical for tests 5 and 6 for both before and after running the bearing. However, micropitting was not produced from test 6.

A fourth bearing was tumbled and run as was performed above. The average ball surface roughness after tumbling was  $0.81 \pm 0.13 \mu\text{m AA}$  ( $32 \pm 5 \mu\text{in. AA}$ ). This surface roughness was higher than that for the previous bearings, even though the tumbling procedure was the same. The bearing was run at 132-percent design maximum radial load (test 7). After running, the bearing was inspected. As with the results of test 6, the bearing had a noticeable wear mark along the load track of the inner race, but micropitting was not evident. The same bearing was used for test 8. The bearing still exhibited no signs of micropitting after testing.

The fifth bearing tested was run at 91 hr with MIL-L-23699 lubricant brand B (test 9). The bearing balls were not tumbled prior to running. After running, the bearing was in very good condition and looked similar to that of the first bearing tested. The last bearing tested was run 100 hr with the DOD-L-85734 lubricant (test 10). The bearing balls were tumbled prior to running, and the average ball surface roughness after tumbling was  $0.61 \pm 0.15 \mu\text{m AA}$  ( $24 \pm 6 \mu\text{in. AA}$ ). After running, the bearing looked similar to those of tests 5 to 7. In summary, the two bearings that had no surface modifications were in good condition after running, with no significant wear on the balls or raceways. Of the four bearings that had surface modifications, one exhibited micropitting. The remaining three had noticeable wear marks on the inner races but without micropitting.

#### Spiral-Bevel Mesh Results

Only slight success was achieved in developing a testing procedure to produce spiral bevel gear scoring failures. The first set of bevel gears tested was unsuccessful in that a fatigue spall was produced on one pinion tooth during the middle of test 2. Other than the spall, both the pinion and the gear showed good contact pattern and no signs of any surface distress. These were the original bevel gears of the transmission.

The second set of bevel gears tested showed signs of heavy wear and light scoring after test 3 (Fig. 8). The set was inspected after test 2 and showed good contact pattern and no surface distress. After test 3 the mesh exhibited heavy wear/light scoring and some micropitting. Heavy wear was evident on the pinion from the root to the tip along most its face width (Fig. 8(a)). The pinion also had small pit lines in the middle of the tooth face and toward the root at the start of contact about 6 to 9 mm wide. The output gear exhibited heavy wear in the middle of the tooth along most its face width (Fig. 8(b)). One concern was the high contact pattern that devel-



(a) Pinion.



(b) Output gear.

FIG. 8.—Spiral bevel gear mesh heavy wear after test 3.

oped on the pinion (pattern toward the tooth tip). The cause of the high pattern is not known.

A third set of bevel gears was tested to attempt to repeat this failure mode. The third set of gears was initially run 50 hr at 100-percent torque and  $121^\circ\text{C}$  oil inlet temperature (test 4) to evaluate if high temperature alone would produce scoring. After running, the gear set was inspected; the contact pattern was good, and no tooth distress was produced. The mesh was then run 75 hr at 117-percent torque and  $121^\circ\text{C}$  oil inlet temperature (test 5). No tooth distress was produced. The mesh was run an additional 75 hr at these conditions (test 6). Again, no tooth distress was produced.

The lubricant flow rate to the bevel mesh was modified in an attempt to promote tooth scoring. The oil jet to the spiral-bevel mesh was modified by plugging one of the two oil jet holes which lubricated the mesh. The modified oil flow rate was 40-percent that of the nominal as measured using an external oil pumping system. After running 25 hr at 40-percent oil flow rate (the last 25 hr of test 6), no signs of tooth distress were evident. The oil jet was further modified by soldering the holes closed and redrilling one of the two jet holes to approximately one-half the original size. This produced an oil flow rate of 21-percent of nominal. The mesh was then run 100 hr at 21-percent oil flow rate and reduced oil level to ensure that the pinion was not submerged in oil (test 7). After running and upon inspection, both the pinion and gear exhibited a thin, elliptical wear mark, but no scoring occurred. The wear marks were on the pinion near the root at the start of contact and on the gear near the tip, both extending the majority of the tooth widths. The gears were run an additional 100 hr at these conditions using MIL-L-23699 lubricant brand B (test 8). No change in appearance occurred on the gears after this test.

A fourth and fifth set of bevel gears were tested at these conditions with MIL-L-23699 lubricant brand B and DOD L-85734

lubricant (tests 9 and 10, respectively). The gears were in good condition after these tests with no evidence of tooth scoring or distress. Thus, a testing procedure to produce spiral-bevel-gear scoring repeatedly was not achieved.

## SUMMARY OF RESULTS

Experimental tests were performed on the OH-58A helicopter main-rotor transmission in the NASA 500-hp helicopter transmission test stand. The objective was to develop a procedure to produce sun gear and planet bearing fatigue failures, mast shaft ball-bearing micropitting, and spiral bevel gear scoring. These were components that failed due to marginal lubrication from Navy field experience. The following results were obtained:

1. Success was achieved in developing a testing procedure to produce sun gear and planet bearing fatigue failures. Three planet bearing fatigue failures occurred using a MIL-L-23699 lubricant. Each failure occurred within 75 hr at 117-percent of design maximum torque and 121 °C (250 °F) oil inlet temperature. Four sun gear fatigue failures occurred using two different brands of MIL-L-23699 lubricant.

2. Only slight success was achieved in developing a testing procedure to produce mast shaft ball-bearing micropitting. Four bearings had surface modifications performed to roughen their surface finishes and promote marginal lubrication. Of these four, one bearing exhibited micropitting when run in endurance tests.

3. A testing procedure to repeatedly produce spiral bevel gear scoring was not achieved. One set out of five exhibited heavy wear and light scoring when run at 117-percent torque and 121 °C oil inlet temperature. Reduced oil flow rate of 21-percent of nominal had very slight effect on tooth surface distress.

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